1998

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ABSTRACT

This paper reviews the design of a combination variable / fixed-step port unloader system for moderate capacity refrigeration screw compressors. The general specification of an unloader system to meet application requirements and specific issues of port and flowpath design are discussed. The paper demonstrates use of a comprehensive compressor simulation and a commercial pipe flow analysis program in the design. Comparisons with experimental data are also presented.

INTRODUCTION

A slide piston / axial port unloading scheme for screw compressors used in multi-circuit refrigeration systems can provide modulated capacity control over a significant portion of the operating load line. For this application, use of such an unloader system has little effect on full load performance. In the following, design requirements, design methods and the selection of details for a specific piston / port system for a screw compressor for air-cooled water chillers are described.

UNLOADER SYSTEM DESCRIPTION

The unloader system consists of a modulating capacity slide piston in the male rotor housing (Figure 1) and a flush axial piston on the female rotor discharge end plane (Figure 2). A complete description of this concept may be found in the U. S. Patent by Linnert /1/.

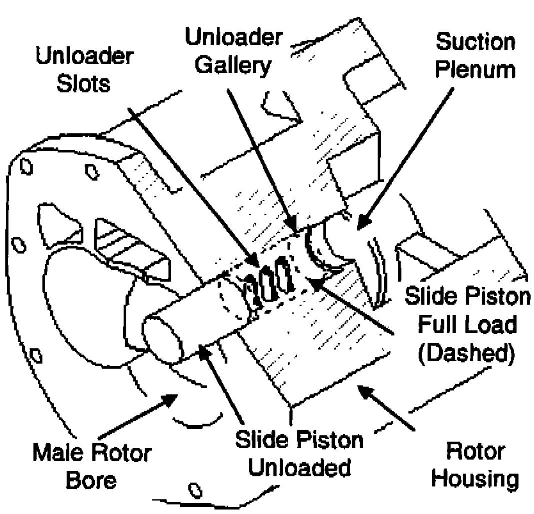

Figure 1 - Slide Piston Unloader Geometry

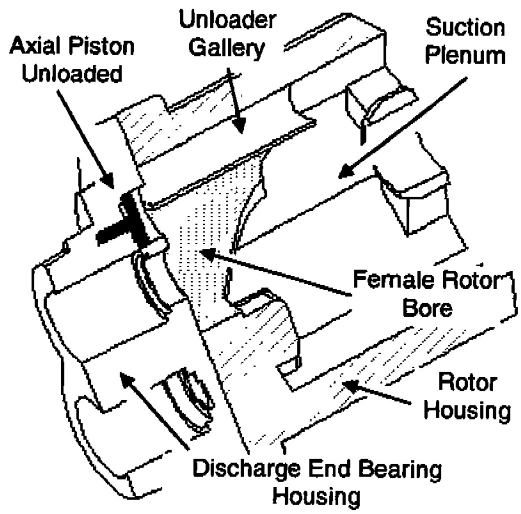

Figure 2- Axial Port Unloader Geometry

The slide piston moves in a bore on the male rotor side which is open at one end to the suction plenum. Compression pockets formed by the rotor lobes connect to the bore through the unloader slots. When the piston is in the full forward position, the slots are blocked and there is no flow to the inlet to reduce capacity. Withdrawing the piston uncovers the slots and refrigerant flows back to the inlet. Helically aligned slots as shown in Figure 1 allow nearly continuous capacity modulation until the last slot is fully uncovered. Further unloading is provided as a step by opening the female side port.

On the female side, the unloader port is closed when the piston is flush with the discharge end plane. When the piston is withdrawn, the compression space between female lobes is connected to the unloader gallery and flow from the rotors is returned to the inlet for capacity reduction. Opening the female side port results in a step change in capacity whose magnitude depends on the location of the port.

DESIGN GOALS AND UNLOADER SYSTEM CHOICES

Design of the unloader system is carried out to meet the following, sometimes conflicting, application and performance goals:

- Reliable
- Little effect on full load performance
- Modulated capacity with no single compressor/dual compressor control dead band region
- Provide 25% of full load system capacity
- Low cost

Slide valve unloading provides continuous capacity modulation. With appropriate selection of slide valve length and axial port volume ratio /2/ it provides efficient part load operation down to 25% load with a single compressor. However, tolerance stack-ups require extra clearance between the rotors and slide valve. This clearance is critical for performance /3/, and is more important for smaller compressors. Slide valves require critical machining tolerances in several parts and thus tend to be relatively costly. The effect of the slide valve on total cost is also more important in smaller compressors.

The piston/port concept is a low cost alternative to the slide valve. There are fewer critical machining dimensions and no extra clearances between rotor and housing are required. This concept does introduce a re-expansion loss from recessed ports and, unless full load performance is compromised by using high volume ratios, a single compressor cannot provide 25% of full load capacity. These disadvantages are moderated when compressors are used in dual-circuit systems. In this case, the slots can be kept in the relatively low pressure range of compression where re-expansion losses are low. The loss of modulated capacity range is offset by using two compressors. A single-compressor capacity modulation range from 100% to about 60% CFM (suction volume flow rate) is typical. The maximum unloading is limited by how close the female side port can be located relative to the discharge ports. Current designs allow unloading to about 35% of the compressor's full load CFM.

The first step in the design process is to set the load steps and operating conditions required. This is an iterative process between compressor and system design. For the purposes of this report, only the results of the analysis are given. Operating conditions and load steps that define unloader design requirements are given in Table 1 for a dual-compressor, R-22, air-cooled water chiller system.

Table 1 - Capacity Step Requirements

Step	***System Load State***	***Compressor 1 Load State****	***Compressor 2 Load State***	***Balance Point Pressure Ratio***
A	full load	full	full	3.72
B	mid load (1)	full	off	3.01
C	mid load (2)	m	m/f	2.57
D	min load	m/f	off	1.98

* Compressor load states

full = no bypass flow through unloader slots/port

m = unloading bypass through all male side slots

m/f = unloading bypass through all male side slots and female side port

The full load state **A** defines compressor displacement. A chiller system analysis provides the minimum load balance point **D** which should have no more than 25% of the full load chiller capacity. For this, one compressor must run at 42% of its full load suction volume flow rate. The capacity reduction achieved in a screw compressor is highly dependent on where in compression the last unloader port is open for connection to suction. Thus, the minimum load requirement determines the location of the female side port, a location relatively independent of where the male side piston slots are placed.

Mid load requirements are set to provide capacity overlap for continuous capacity control in the transition from single- to dual-compressor operation. To meet this requirement, total compressor CFM for load state **C** must be less than the CFM of a single compressor at state **B**. It is this requirement that determines the location of the male rotor slide piston ports. The number and spacing of these ports is determined based on unloaded performance and achieving continuous capacity modulation.

With capacity requirements known, the task is to define design details in terms of port sizes and locations, configuration of male side slots and unloader gallery geometries that provide the necessary volume flows, operate efficiently at the part load states that have minimal effect on full load performance.

DESIGN APPROACH

The compressor simulation described in /4/ allows for simple analysis of the piston / port unloader and is used to size and locate the unloader elements. In the ports connecting the rotors to the unloader galleries, the flow is modeled as isentropic, modified by an estimated or empirically determined flow coefficient. Figure 3 and equations 1 summarize the calculations. The rotor side state point (c) is defined by the instantaneous conditions computed by the compression process models. The gallery conditions (g) are computed from the unloader gallery performance. The program uses these conditions to compute the stagnation and static enthalpies in the port cross section (p) from which a velocity can be computed. This velocity, static density and port area combine to determine the isentropic mass flow rate.

$$
\left.
\begin{aligned}
&H_{0c} = f(P_{0c}, T_{0c}) && S_c = S_p = f(P_{0c}, T_{0c}) \\
&H_{0p} = H_{0c} && \\
&P_p = P_g && H_p = f(P_p, S_p) \\
&\rho_p = f(P_p, S_p) && \\
&V_p^2 = 2 \cdot (H_{0p} - H_p) && \dot{m}_p = \Psi_p \cdot \rho_p \cdot V_p \cdot A_p
\end{aligned}
\right| \quad (1)
$$

where:

H - enthalpy	ρ - density	ṁ - mass flow rate
P - pressure	S - entropy	Ψ - flow coefficient
T - temperature	V - velocity	

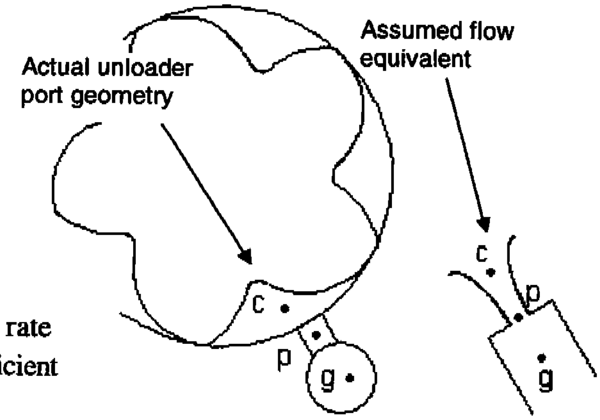

Figure 3 - Port Flow Model

In equations 1, the subscripts c, p and g refer to conditions in the compression chamber, port and gallery as shown in the figure. Subscript 0 on a property denotes a stagnation state.

The unloader gallery pressure P_g is assumed to be constant and equals the rotor inlet pressure plus a pressure loss. The gallery pressure loss is computed by specifying a reference flow path area and a total pressure loss coefficient, DPG, defined as the gallery pressure loss referred to the dynamic pressure at the reference area, A_g. The dynamic pressure is in turn computed from the gallery mass flow rate $\dot{m}_g$, which is the sum of the individual port flows, $\dot{m}_p$. The calculation approach, which applies to either the male or female side galleries, is outlined in equations 2.

$$DPG = \frac{\Delta P_g}{q_g} \qquad \text{so that} \qquad P_g = P_{inlet} + DPG \cdot q_g$$

where

$$q_g = \frac{\rho_g \cdot V_g^2}{2} \qquad \text{and} \qquad V_g = \frac{\dot{m}_g}{\rho_g \cdot A_g} \qquad (2)$$

The male side unloader geometry causes an additional compression loss at full load. The recessed passage between the compression chamber and unloader gallery collects compressed vapor which is eventually re-expanded into a lower pressure chamber. This effect was studied during a 1989 research project where several port configurations were tested. Tests included changing the number and size of ports, studying use of ports on the male and female side or the male side only, and varying the location of the ports in the compression process. These tests demonstrated the effect of the ports on full load performance and provided information to calibrate the model in the simulation program. A complete description of this effort is beyond the scope of the present report. However, two important results are summarized below.

- The research work showed that an unloader scheme using only slide piston ports arranged to provide the required minimum 42% suction volume flow resulted in a 3.8% loss in full load performance. Other configurations tested revealed that the slide piston scheme could be used to unload the compressor to 60% of full load CFM with less than a 1% change in full load performance.
- A modification to the radial leakage analysis in the compressor simulation was added to account for the effect of the slide piston ports. When a rotor lobe is passing over a port, a factor is added to the radial leakage area. This additional area depends on port size (diameter of a round port or width and length of a slot). While this was viewed as an interim model, it proved to be reliable in predicting full load losses after calibration with only two test configurations and has therefore not been replaced in the simulation program. An alternative analysis method where the re-expansion loss is considered directly may be found in the paper on scroll compressor design by Lifson, et. al. /5/.

Port flow coefficients Ψ and gallery loss coefficients DPG were originally found experimentally. Recently, a general purpose pipe flow analysis program, Fathom™ from Applied Flow Technology Corp., has been used to analyze designs during early stages of configuration determination. This program allows construction of flow networks using pre-defined models for pipes and piping system elements such as orifices, expansions, contractions and turns. Figure 4 shows a network constructed for analysis of a slide piston unloader in a production compressor whose performance had been previously determined experimentally. The original calibration for the port flow coefficient and unloader gallery loss coefficient resulted in values of $\Psi = 0.67$ and DPG = 1.05. The Fathom™ analysis resulted in values of $\Psi = 0.73$ (average of three ports; variation was ± 2.8%) and DPG = 1.60. The same analysis was applied to the female side port and unloader gallery flowpath, resulting in computed port flow and gallery loss coefficients of 0.80 and 2.20, respectively.

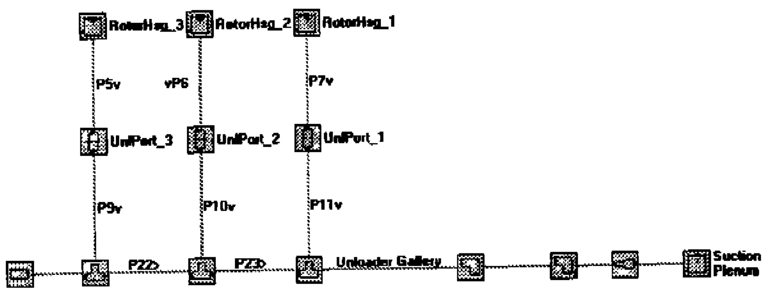

Figure 4 - Fathom™ Model of Slide Piston Unloader

DESIGN EXAMPLE

A slide piston / axial port unloader system was designed for a nominal 60 ton, R-22 compressor for use in dual-compressor, air-cooled water chillers. This compressor has a 5 lobe male rotor with 127mm diameter, 250° wrap angle and 1.35 L/D. The 7 lobe female is 117mm in diameter. For the air-cooled application, the compressor uses a 3.1 volume ratio.

Details of the unloader geometry are shown in Table 2, derived from the simulation program output.

Table 2 - Unloader Geometry Specifications for Simulation Program

```
PORT UNLOADER DATA
   UNLOADER NUMBER                  1       2       3       4
   UNLOADER TYPE               RADIAL  RADIAL  RADIAL   AXIAL
   UNLOADER SIDE                    M       M       M       F
   SLOT WIDTH  (IN.)              .30     .30     .30     .00
   SLOT LENGTH (IN.)              .80     .80     .80     .00
   VI at Opening                 1.05    1.16    1.33    1.30
   VI at Closure                 1.57    1.91    2.43    3.84
   FULLY OPEN AREA (IN**2)      .2205   .2205   .2205  1.0400
   FLOW COEFFICIENT             .7300   .7300   .7300   .8000
   CLEARANCE FACTOR             .1200   .1200   .1200   .0000
```

The effect of the location of the male side slide piston slots on the minimum load CFM achieved when the female side port is opened is small, but in the final configuration selection, several female port locations must be analyzed to achieve the desired results. In the design of the 127mm compressor, the female unloader port is in a location such that there is a period of time where the compression chamber is open to both the axial unloader port and the 3.1 volume ratio discharge port. Table 2 shows the unloader port is not closed until a geometric compression of 3.84 volume ratio. This is acceptable since during the overlap period, the open areas to both the unloader and discharge ports are small. In addition, the system is designed with sufficient oil flow and oil cooling to control discharge temperatures, even at higher operating pressure ratios. Table 3 compares the calculated and actual capacities at the design load steps given in Table 1 for the unloader set of Table 2.

Table 3- Computed Capacities

Step	System Load State	Compressor 1 CFM Calculated / Actual	Compressor 2 CFM Calculated / Actual	Total CFM % of Full Load Calculated / Actual
A	full load	121.7 / 121.6	121.7 / 121.6	100 / 100
B	mid load (1)	125.5 / 125.7	0 / 0	51.6 / 51.7
C	mid load (2)	73.4 / 72.9	38.8 / 41.7	46.1 / 47.3
D	min load	42.4 / 42.9	0 / 0	17.4 / 17.6

The minimum load CFM for this design allows operation at about 21% of full load capacity. In addition, the mid load characteristics satisfy the control overlap requirement that the total compressor CFM for step **C** be less than that for step **B**.

The effect of the three male side slots, based on computations using calibrations from the previously mentioned research work, was an 0.9% reduction in full load efficiency.

SENSITIVITY STUDY EXAMPLE

The usefulness of any design model is in its application to studying the impact of design choices on performance. For the piston / port unloader, there are numerous parameters whose values will effect full load performance, part load performance or both. The size and shape of the slots, the size of the axial piston and design details of the unloader gallery are all important. Use of the modeling system will be illustrated here by demonstrating the impact of the slide piston side unloader gallery. In the design documented in the previous section, the main gallery channel diameter was 1.12 inches and the overall gallery loss coefficient was 1.60. For this example, the passage diameter is reduced to 0.88 inches. Running this geometry through the Fathom™ program gives an overall loss coefficient of 1.53. This gallery is analyzed with the simulation program at the state **C**, compressor 1 load state (Table 2) with the remainder of the unloader geometric features the same. The higher pressure loss in the gallery (the reference section dynamic pressure upon which the loss coefficient acts increases by 62%, more than offsetting the 4.5% lower loss coefficient) means lower flow through the ports and less unloading. The simulation showed that the smaller gallery area resulted in 5.7% higher capacity. To achieve the target CFM for the state **C** operating point, the unloader slots had to be moved 0.2 inches farther into the compression. In this case, while the CFM is at the target value, calculated compression power is 4.8% higher than with the 1.12" gallery. The higher power is easily seen in the computed pressure-volume diagrams shown in Figure 5. The extra work in the area between the curves for the 1.12" and 0.88" galleries comes from the higher gallery pressure against which the compression pocket must discharge and from the higher level of initial work caused by moving the ports farther into the compression process. In addition to the loss in performance at this particular operating condition caused by the change in the unloader gallery, the need to move the ports into a higher pressure region resulted in a small loss in full load performance (about 0.1%).

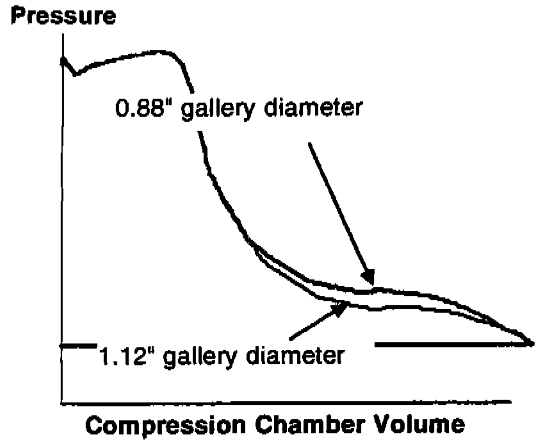

Figure 5 - Computed Effect of Gallery Area on the Compression Process

CONCLUSIONS

The slide piston / axial port unloader system is an alternative to slide valve unloading control that offers low cost and high reliability in small screw compressors for application in air- and water-cooled water chillers. Disadvantages of this concept are a loss in full load performance, a smaller range of modulated capacity and a higher minimum load. These disadvantages are overcome when compressors with the piston / port unloaders are used in pairs.

Experience has shown that reliable design decisions can be made using relatively simple simulation models. When key coefficients in the models are determined from experiment or more sophisticated analyses, the simulation also gives good absolute performance predictions. Finally, sensitivity studies demonstrate the need for careful design of all of the elements in the unloader flowpath.

ACKNOWLEDGMENTS

The author wishes to thank the Trane Company for providing the time to prepare this paper and for the permission to publish the information. My thanks also to the International Compressor Engineering Conference organizing committee for their efforts in providing this forum.

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